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Sugiura

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(54) **VALVE TIMING ADJUSTMENT APPARATUS**

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F01L 1/352 (2006.01)

(52) **U.S. Cl.**
CPC ***F01L 1/352*** (2013.01); *F01L 2250/02* (2013.01); *F01L 2810/02* (2013.01); *F01L 2820/032* (2013.01)

(58) **Field of Classification Search**
CPC . F01L 1/352; F01L 2820/032; F01L 2810/02; F01L 2250/02
USPC 123/90.17
See application file for complete search history.

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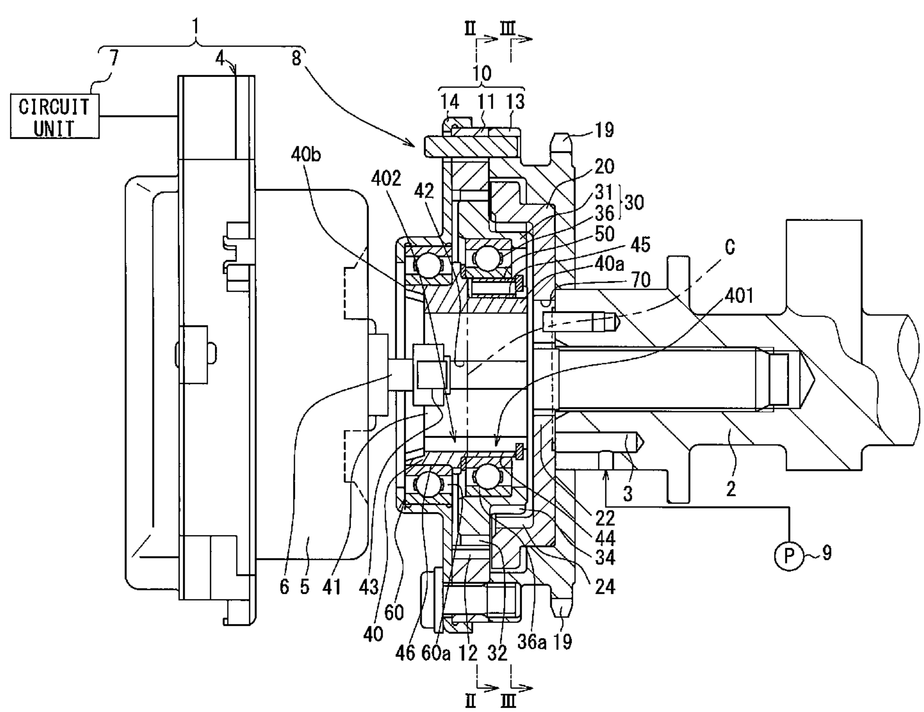
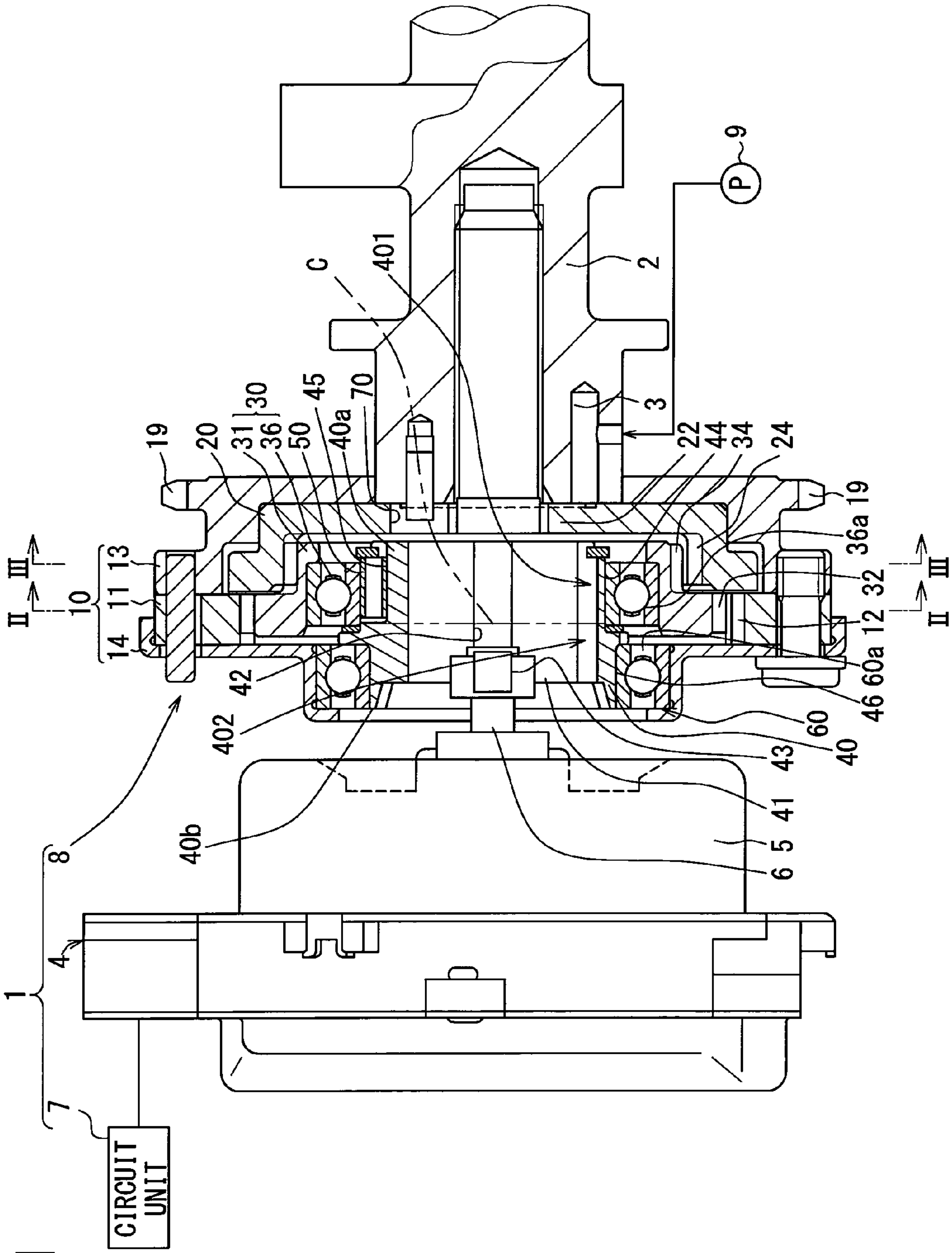
(57) **ABSTRACT**

A planetary carrier supports a geared rotor from an inside in a radial direction, and receives a radial load in a first region, which is offset to one end from a center position in an axial direction. A ball bearing roller supports the planetary carrier from an outside in the radial direction in a second region, which is offset to an other end from the center position. The ball bearing roller is a single-row ball bearing roller that has an outer ring supported by a drive rotor, an inner ring which supports the planetary carrier, and a plurality of rolling elements which are rotatably installed in a single row to be in contact with and between the outer and the inner rings. The geared rotor is tilted relative to the axial direction, and contacts a driven rotor in the axial direction. An angle of the geared rotor relative to the axial direction is set to be smaller than a maximum allowable angle at which the inner ring is allowed to be tilted relative to the axial direction.

5 Claims, 7 Drawing Sheets




CIRCUIT UNIT
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36a
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60a
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60
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43
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40b
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III
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FIG. 1

FIG. 2

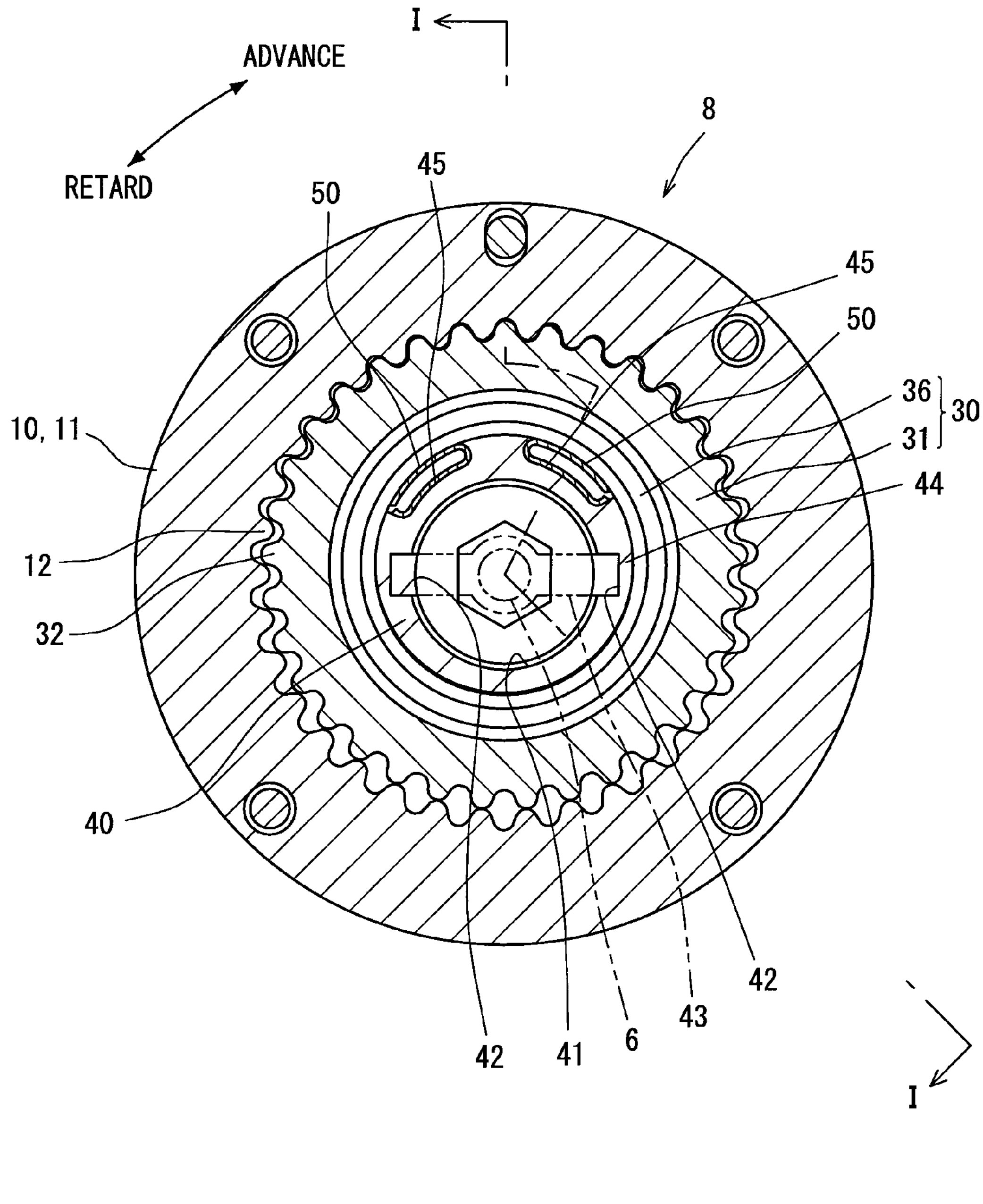
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FIG. 3

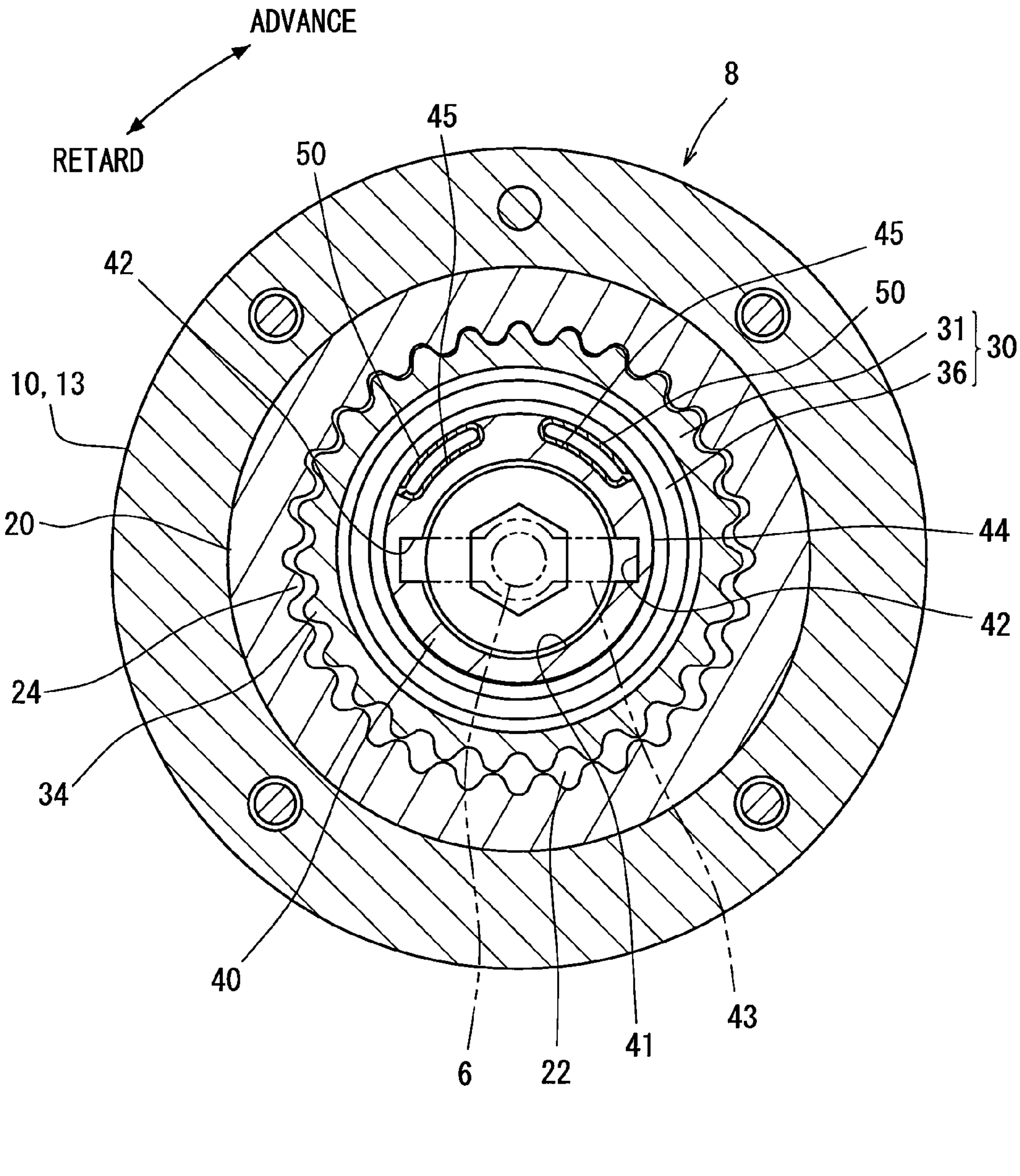
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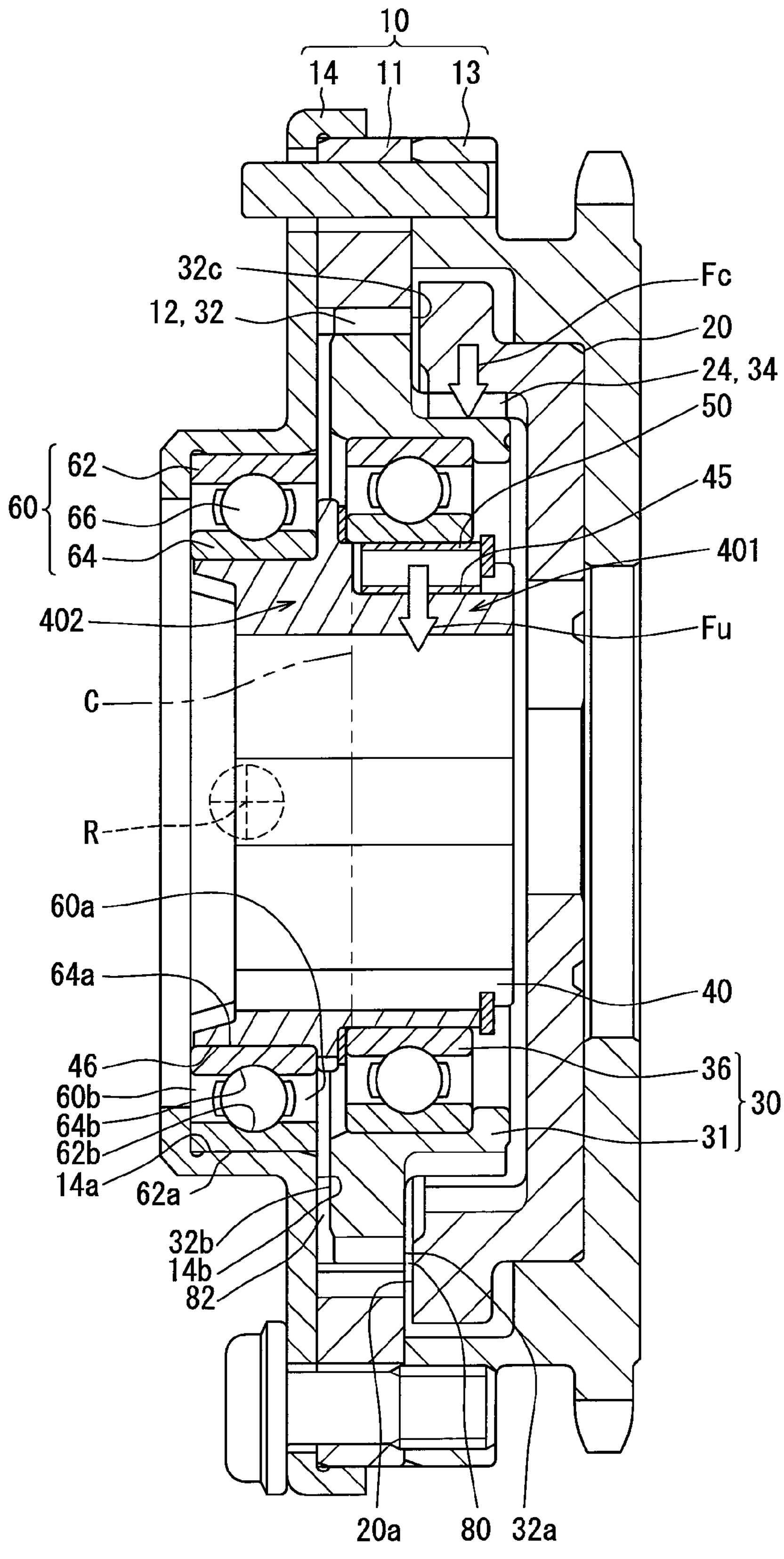
FIG. 4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401
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60a
64a
40
46
60b
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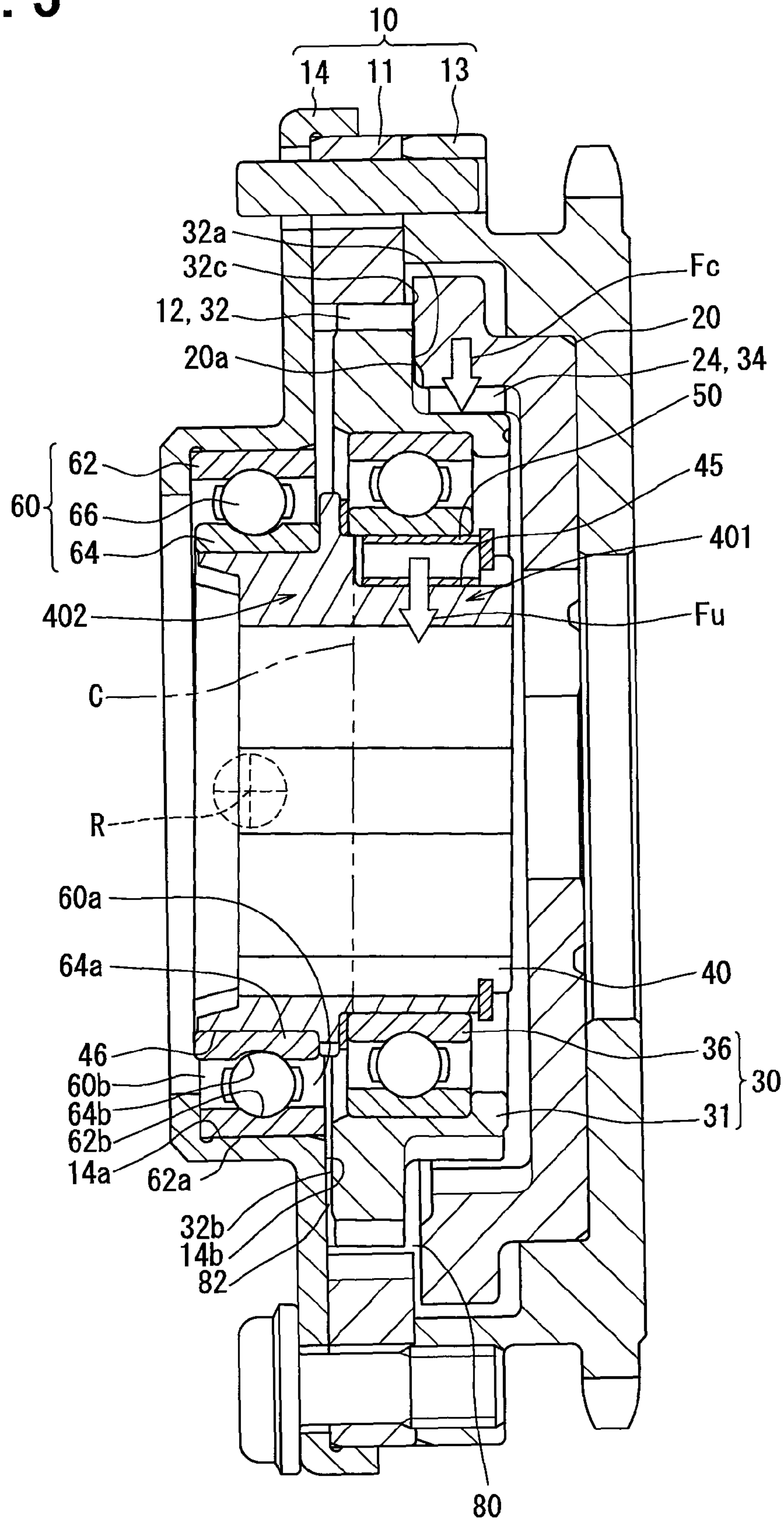
14a
62a
36
30
31
32b
14b
82
20a
80
32a

FIG. 5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C
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14b
82
80

FIG. 6

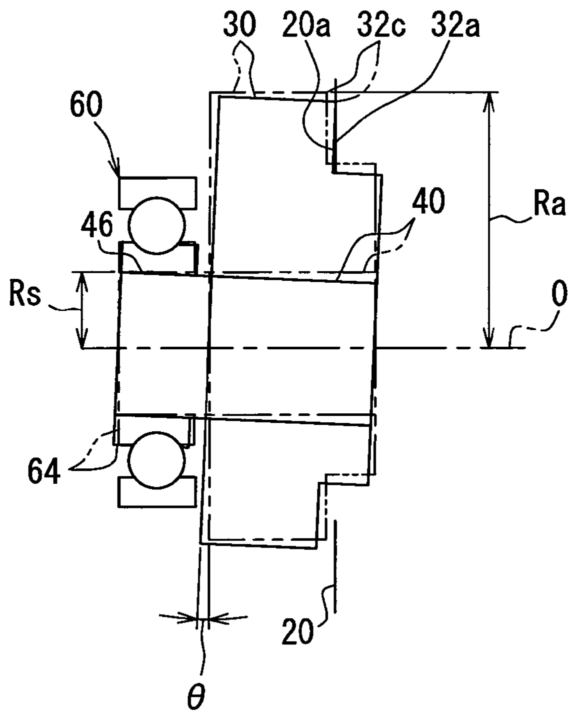


FIG. 7

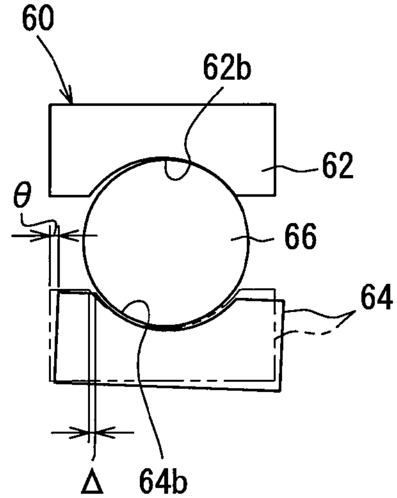
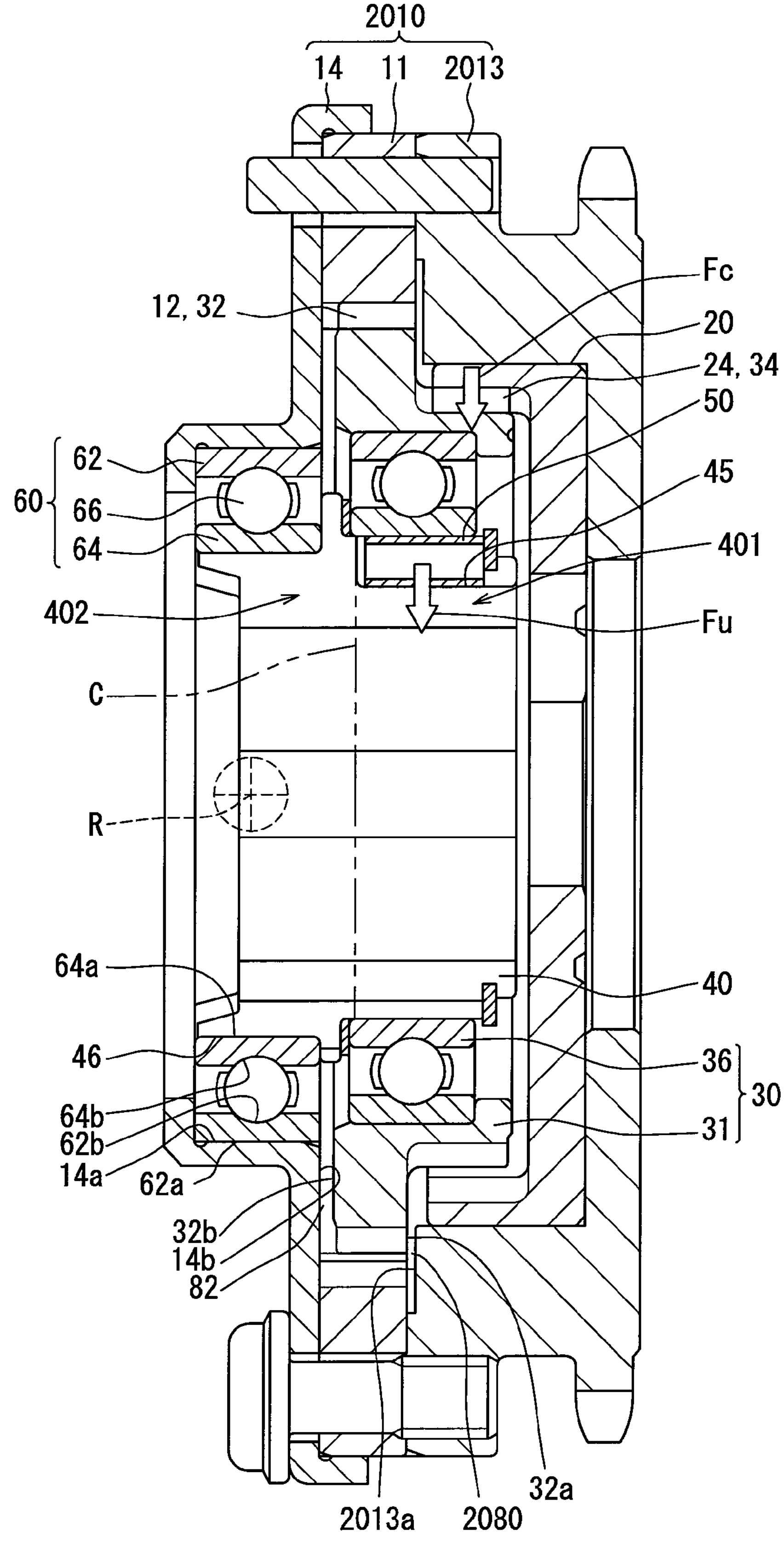


FIG. 8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2013a
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VALVE TIMING ADJUSTMENT APPARATUS

CROSS REFERENCE TO RELATED APPLICATION

This application is based on and incorporates herein by reference Japanese Patent Application No. 2013-125970 filed on Jun. 14, 2013.

TECHNICAL FIELD

The present disclosure relates to a valve timing adjustment apparatus that adjusts valve timing of a valve which is opened and closed by a cam shaft by transmission of a torque from a crank shaft in an internal combustion engine.

BACKGROUND

In the related art, a valve timing adjustment apparatus adjusts a relative phase (hereinafter, referred to as a "rotor-to-rotor phase") between first and second rotors. The first and second rotors are coupled with a crank shaft and a cam shaft through a planetary motion of a geared rotor and respectively rotate.

In a type of the apparatus disclosed in JP 4442574 B, a planetary carrier supports the geared rotor from the inside in a radial direction, which meshes with a first gear portion of the first rotor and a second gear portion of the second rotor. In particular, the first rotor is supported from the outside in the radial direction by ball bearing rollers which are mounted on the second rotor to interpose the first rotor on both sides in an axial direction. As a result, the planetary carrier is tilted relative to the axial direction within a range of an internal clearance of the bearing.

In the apparatus disclosed in JP 4442574 B, the planetary carrier receives a radial load in a first region that is offset in the axial direction to one end from a center position, and the planetary carrier is supported by the ball bearing rollers in a second region that is offset to the other end from the center position. Accordingly, when the planetary carrier receives the radial load in the first region, the planetary carrier is tilted around a specific point as a center point in the second region. The geared rotor supported by the planetary carrier is tilted together with the planetary carrier, and thus the geared rotor undergoes increasing wear due to gear rattle occurring at locations at which the geared rotor meshes with the first and the second gear portions. Accordingly, from the viewpoint of durability the amount of tilt is required to be regulated.

Herein, in the ball bearing roller of the apparatus disclosed in JP 4442574 B, a plurality of rolling elements arranged in two rows are installed between an outer ring mounted on the second rotor and an inner ring that supports the planetary carrier. With the plurality of rows of rolling elements, when the inner ring and the planetary carrier is tilted relative to the axial direction, the tilting of the inner ring is received by the outer ring through each rolling element in two circumferences that are apart in the axial direction from each other. Effects of tilt regulation by the stop structure become great. However, since either a total contact area between the outer ring and the entirety of the rolling elements or a total contact area between the inner ring and the entirety of the rolling elements becomes large, a contact pressure occurs in a large area of a contact interface when the tilt is regulated. As a result, wear increases in the ball bearing roller, the inner ring of which is tilted, thereby causing a reduced lifespan and reduced durability.

SUMMARY

The present disclosure is made in light of the problems described above, and an object of the present disclosure is to provide a valve timing adjustment apparatus with high durability.

In an aspect of the present disclosure, a valve timing adjustment apparatus is for adjusting valve timing of a valve that is opened and closed by a rotation of a cam shaft. The cam shaft is rotated by torque transmitted from a crank shaft in an internal combustion engine.

The valve timing adjustment apparatus includes a first rotor having a first gear portion and rotatably coupled with one of the crank shaft or the cam shaft, a second rotor having a second gear portion and interposing the first rotor on both sides in an axial direction, the second rotor rotatably coupled with an other of the crank shaft or the cam shaft, and a geared rotor meshing with the first gear portion and the second gear portion and moving in planetary motion to adjust a relative phase between the first rotor and the second rotor. The apparatus further includes a planetary carrier supporting the geared rotor in a radial direction from an inside of the geared rotor, and receiving a radial load in a first region which is offset to one end from a center position of the planetary carrier in the axial direction, and a ball bearing roller that is supported by the second rotor and supports the planetary carrier in the radial direction from an outside of the planetary carrier in a second region which is offset to an other end from the center position of the planetary carrier in the axial direction.

The ball bearing roller is a single-row ball bearing roller that has an outer ring supported by the second rotor, an inner ring that supports the planetary carrier, and a plurality of ball bearings that are rotatably installed in a single row to be in contact with and between the outer ring and the inner ring.

The geared rotor is tilted relative to the axial direction and contacts the first rotor or the second rotor in the axial direction.

An angle of the geared rotor relative to the axial direction is set to be smaller than a maximum allowable angle at which the inner ring is allowed to be tilted relative to the axial direction.

According to an aspect of the present disclosure, the planetary carrier receives the radial load in the first region that is offset to one end from the center position in the axial direction. The ball bearing roller supports the planetary carrier in the second region that is offset to the other end from the center position. Accordingly, when the planetary carrier receives the radial load in the first region, the planetary carrier is angled around a specific point as a center point in the second region.

Here, the ball bearing roller has a structure in which the plurality of rolling elements are installed in only a single row between the outer ring mounted on the second rotor and the inner ring that supports the planetary carrier. Accordingly, in the single-row ball bearing roller, when the planetary carrier and the inner ring are tilted relative to the axial direction, the tilt of the inner ring is received by the outer ring through the rolling elements in a single circumference, and thus effects of tilt regulation becomes small. At this time, the geared rotor supported from the inside in the radial direction by the planetary carrier also is tilted relative to the axial direction, and thus the geared rotor is brought into contact with the first rotor or the second rotor that supports the first rotor. Since the effects of tilt regulation can be realized by the contact, the geared rotor is unlikely to undergo gear rattle at the locations where the geared rotor meshes with the first and the second gear portions. At this time, the geared rotor is tilted together

with the planetary carrier and the inner ring at the angle which is smaller than the maximum allowable angle of the inner ring. When the angle of the geared rotor is small, even in the single-row ball bearing roller in which both a total contact area between the outer ring and the entirety of the rolling elements and a total contact area between the inner ring and the entirety of the rolling elements are small, a contact pressure occurring in a small area of a contact interface can be made small.

In this manner, wear can be prevented from occurring at the locations where the sloping geared rotor meshes with each gear portion. Wear can be also prevented from occurring in the ball bearing roller, the inner ring of which is tilted, thereby being able to improve durability of the entire apparatus.

In another aspect of the present disclosure, the geared rotor is eccentric with respect to the first rotor and the second rotor, and an outermost circumferential angled portion in on eccentric side of the geared rotor is brought into elastic contact with the first rotor or the second rotor. The planetary carrier has a supporting surface coaxial with the first rotor and the second rotor and supports the inner ring with the supporting surface. A radius between a rotational center line of the first rotor and the second rotor and the outermost circumferential angled portion is set to be greater than a radius between the rotational center line and the supporting surface.

According to the aspect of the present invention, the radius between the rotational center line and the outermost circumferential angled portion is set to be greater than the radius between the rotational center line and the supporting surface of the planetary carrier. The outermost circumferential angled portion on the eccentric side of the geared rotor is brought into elastic contact with the first rotor or the second rotor due to the eccentricity of the geared rotor with respect to the first and the second rotors. Accordingly, a large contact area can be ensured, and thus the contact pressure can be suppressed. As a result, wear of the contact interface can be suppressed. In contrast, in the single-row ball bearing roller, the inner ring is supported by the supporting surface, which has the radius from the rotational center line smaller than the radius between the rotational center line and the outermost circumferential angled portion of the geared rotor. Therefore, when the inner ring is tilted together with the planetary carrier, the amount of thrust which causes the inner ring to be offset with respect to the outer ring can be reduced. Here, in particular, in the contact interface between the outer ring and the rolling elements and the contact interface between the inner ring and the rolling elements, wear caused by the offset of the inner ring can be suppressed by the reduction of the amount of thrust. The aforementioned configuration can contribute to an improvement in durability.

BRIEF DESCRIPTION OF THE DRAWINGS

The disclosure, together with additional objectives, features and advantages thereof, will be best understood from the following description, the appended claims and the accompanying drawings, in which:

FIG. **1** is a view illustrating a valve timing adjustment apparatus according to First Embodiment, and is a cross-sectional view taken along line I-I in FIG. **2**;

FIG. **2** is a cross-sectional view taken along line II-II in FIG. **1**;

FIG. **3** is a cross-sectional view taken along line III-III in FIG. **1**;

FIG. **4** is a cross-sectional view illustrating enlarged main portions in FIG. **1**;

FIG. **5** is a cross-sectional view illustrating an operation state different from that in FIG. **4**;

FIG. **6** is a schematic view describing characteristics of the main portions in FIG. **1**;

FIG. **7** is another schematic view describing the characteristics of the main portions in FIG. **1**; and

FIG. **8** is an enlarged cross-sectional view illustrating main portions of a valve timing adjustment apparatus according to Second Embodiment.

Hereinafter, a plurality of embodiments of the present disclosure will be described with reference to the accompanying drawings. In each embodiment, the same reference signs are assigned to corresponding configuration elements, and there is a case where duplicated descriptions are omitted. In each embodiment, when only a part of a configuration of an embodiment is described, a corresponding configuration of another embodiment, which is previously described, is applicable to the other part of the configuration of the embodiment. Insofar as there are no problems with a combination of the configurations, not only can the configurations be combined together as stated in each embodiment, but also the configurations of the plurality of embodiments can be partially combined together even though the partial combinations of the configurations are not stated.

(First Embodiment)

As illustrated in FIG. **1**, a valve timing adjustment apparatus **1** according to First Embodiment of the present disclosure is installed in a transmission system that transmits a crank torque from a crank shaft (not illustrated) to a cam shaft **2** in an internal combustion engine of a vehicle. In the embodiment, the cam shaft **2** opens and closes an intake valve (not illustrated) among "valves" of the internal combustion engine through the transmission of the crank torque, and the apparatus **1** adjusts valve timing of the intake valve.

(Basic Configuration)

Hereinafter, a basic configuration of the apparatus **1** will be described. As illustrated in FIGS. **1** to **3**, the apparatus **1** is configured to have an actuator **4**, an electrification control circuit unit **7**, a phase adjustment unit **8** and the like.

For example, the actuator **4** which is illustrated in FIG. **1** is an electrically driven brushless motor, and has a housing body **5** and a control shaft **6**. The housing body **5** is fixed to a fixed joint of the internal combustion engine, and the housing body **5** supports the control shaft **6** that can rotate in both circumferential directions (in a clockwise direction and in a counterclockwise direction in FIGS. **2** and **3**). For example, the electrification control circuit unit **7** is configured to have a driving driver, a micro computer for control of the driving driver and the like. The electrification control circuit unit **7** is disposed outside and/or inside the housing body **5**. The electrification control circuit unit **7** drives a rotation of the control shaft **6** by controlling electrification of the actuator **4** that is electrically connected to the electrification control circuit unit **7**.

The phase adjustment unit **8** includes a drive rotor **10**, a driven rotor **20**, a geared rotor **30**, a planetary carrier **40**, an elastic member **50** and a ball bearing roller **60**.

As illustrated in FIGS. **1** to **3**, the hollow metallic drive rotor **10** accommodates the other configuration elements **20**, **30**, **40**, **50** and **60** of the phase adjustment unit **8** therein. The drive rotor **10** tightens a sun gear member **11**, a sprocket member **13** and a cover member **14** together in a state where the sun gear member **11** is interposed between the sprocket member **13** and the cover member **14**.

As illustrated in FIGS. **1** and **2**, a drive inner gear portion **12** is formed on an inner circumferential surface of a circumferential wall portion of the annular plate-shaped sun gear member **11**, and the drive inner gear portion **12** has an addendum circle which is present inward from a root circle in a radial direction. As illustrated in FIG. **1**, a plurality of sprocket teeth **19** are formed on an outer circumferential surface of a circumferential wall portion of the bottomed cylindrical sprocket member **13**, and the sprocket teeth **19** protrude outward from locations of the sprocket member **13**, which are apart from each other at equal intervals in the circumferential direction. A timing chain (not illustrated) is suspended between the sprocket teeth **19** and a plurality of sprocket teeth of the crank shaft, and thus the sprocket member **13** is connected to the crank shaft. When the crank torque of the crank shaft is transmitted to the sprocket member **13** through the timing chain, the connection allows the drive rotor **10** to rotate in a constant circumferential direction (in the clockwise direction in FIGS. **2** and **3**) in conjunction with the crank shaft.

As illustrated in FIGS. **1** and **3**, the bottomed cylindrical metallic driven rotor **20** is coaxially fitted into the sprocket member **13**, and thus the drive rotor **10** is supported from the inside in the radial direction by the driven rotor **20**. The driven rotor **20** is interposed in an axial direction between the sun gear member **11** and the sprocket member **13**. A connection portion **22** is formed on a bottom wall portion of the driven rotor **20** to be coaxially connected to the cam shaft **2**. The connection allows the driven rotor **20** to rotate in the same circumferential direction (in the clockwise direction in FIG. **3**) as that of the drive rotor **10**, and the driven rotor **20** can rotate relative to the drive rotor **10** in both circumferential directions.

A driven inner gear portion **24** is formed on an inner circumferential surface of a circumferential wall portion of the driven rotor **20**, and the driven inner gear portion **24** has an addendum circle which is present inward from a root circle in the radial direction. The driven inner gear portion **24** is disposed to be offset in the axial direction from the drive inner gear portion **12**. The driven inner gear portion **24** is set to have an inner diameter smaller than that of the drive inner gear portion **12**. The driven inner gear portion **24** is set to have the number of teeth less than that of the drive inner gear portion **12**.

As illustrated in FIGS. **1** to **3**, the metallic geared rotor **30** is disposed in the radial direction from the inside of the sprocket member **13** and the driven rotor **20** to the inside of the sun gear member **11**. The geared rotor **30** is an assembly of a planetary geared member **31** and a planetary bearing **36**.

The stepped annular plate-shaped planetary geared member **31** is eccentrically disposed with respect to the rotors **10**, **20** and the control shaft **6**. A drive outer gear portion **32** and a driven outer gear portion **34** are formed on an outer circumferential surface of a circumferential wall portion of the planetary geared member **31**. Each of the drive outer gear portion **32** and the driven outer gear portion **34** has an addendum circle which is present inward from a root circle in the radial direction. The drive outer gear portion **32** meshes with the drive inner gear portion **12** in such a manner that a meshing position of the drive outer gear portion **32** is eccentrically located in the radial direction with respect to the rotors **10** and **20**. The driven outer gear portion **34** is disposed to be offset in the axial direction from the drive outer gear portion **32**. The driven outer gear portion **34** is set to have an outer diameter smaller than that of the drive outer gear portion **32**. The driven outer gear portion **34** is set to have the number of teeth less than that of the drive outer gear portion **32**. The driven outer gear portion **34** meshes with the driven inner gear portion **24** in such a manner that a meshing position of the driven outer gear portion **34** is eccentrically located in the radial direction with respect to the rotors **10** and **20**.

In the embodiment, the planetary bearing **36** is a single-row ball bearing roller, and is eccentrically disposed with respect to the rotors **10** and **20** and the control shaft **6**. The planetary bearing **36** is coaxially fitted into the planetary geared member **31**, and thus the planetary bearing **36** is retained by the planetary geared member **31**.

The partially eccentric cylindrical metallic planetary carrier **40** is disposed in the radial direction from the inside of the sprocket member **13** and the driven rotor **20** to the inside of the cover member **14**. A cylindrical input surface **41** is formed on an inner circumferential surface of a circumferential wall portion of the planetary carrier **40**. The input surface **41** is coaxial with the rotors **10** and **20** and the control shaft **6**. The input surface **41** is provided with connection grooves **42** that are fitted into a coupler **43**, and the control shaft **6** is connected to the planetary carrier **40** through the coupler **43**. The connection allows the planetary carrier **40** to rotate in the circumferential direction integrally with the control shaft **6**, and the planetary carrier **40** can rotate in both circumferential directions relative to the drive inner gear portion **12**.

A cylindrical eccentric surface **44** is formed on an outer circumferential surface of the circumferential wall portion of the planetary carrier **40**, and the eccentric surface **44** is eccentric with respect to the rotors **10** and **20** and the control shaft **6**. The eccentric surface **44** is provided across the following two regions in the planetary carrier **40**: a first region **401** that is offset to one end **40***a* from a center position C in the axial direction, and a second region **402** that is offset to the other end **40***b* from the position C. In practicality, the planetary bearing **36** is coaxially fitted onto the eccentric surface **44** in the first region **401**, and thus the planetary bearing **36** is installed in the radial direction between the planetary carrier **40** and the planetary geared member **31**. The installation allows the planetary carrier **40** to support the geared rotor **30** from the inside in the radial direction, and thus the geared rotor **30** can move in planetary motion. The planetary motion is defined as such a motion that the geared rotor **30** rotates in the circumferential direction, and revolves in a rotational direction of the control shaft **6** and the planetary carrier **40**.

Here, the side toward which the respective outer gear portions **32** and **34** are eccentric with respect to the inner gear portions **12** and **24** (the upper side of FIG. **1**) substantially coincide with the side toward which the eccentric surface **44** is eccentric with respect to the rotors **10** and **20** and the control shaft **6**, and hereinafter, a side toward which a portion is eccentric with respect to another portion is simply referred to as an "eccentric side". A cam torque transmitted from the cam shaft **2** to the driven rotor **20** by a reaction force of an intake valve spring is converted into a force based on a pressure angle at locations at which the gear portions **24** and **34** mesh with each other as illustrated in FIG. **4**. Accordingly, a radial load Fc occurs toward a direction opposite to the eccentric side. As a result, the eccentric surface **44** of the planetary carrier **40** receives the radial load Fc in the first region **401**.

As illustrated in FIG. **1**, a cylindrical supporting surface **46** is formed on the outer circumferential surface of the circumferential wall portion of the planetary carrier **40**, and the supporting surface **46** is coaxial with the rotors **10** and **20** and the control shaft **6**. The supporting surface **46** is provided in the second region **402** that is offset by the planetary carrier **40** to the other end **40***b* from the center position C in the axial direction.

As illustrated in FIGS. **1** to **3**, the metallic elastic member **50** is individually accommodated in each of accommodating holes **45** that are open at two circumferential locations of the eccentric surface **44** in the first region **401**. Each elastic member **50** is a leaf spring generally with a U-shaped cross section. A resultant force of a biasing force (a restoring force) caused by an elastic deformation of each elastic member **50** is exerted on the geared rotor **30** toward the eccentric side while the resultant force is exerted on the planetary carrier **40** in the direction opposite to the eccentric side. As a result, as illustrated in FIG. **4**, the resultant force of the biasing force of each elastic member **50** becomes the radial load that each of the inner gear portions **12** and **24** receives at the locations at which the inner gear portions **12** and **24** mesh with the outer gear portions **32** and **34**, respectively. The resultant force becomes a radial load Fu that an inner bottom surface of the accommodating hole **45** receives in the first region **401**.

As illustrated in FIGS. **1** and **4**, the single-row metallic ball bearing roller **60** is coaxially fitted into a cylindrical portion of the hat-shaped cover member **14**, and thus the ball bearing roller **60** is retained by the drive rotor **10**. The ball bearing roller **60** is coaxially fitted onto the supporting surface **46**, and is installed in the radial direction between the planetary carrier **40** and the drive rotor **10**. The installation allows the ball bearing roller **60** to rotatably support the planetary carrier **40** from the outside in the radial direction.

The phase adjustment unit **8** with the aforementioned configuration adjusts a phase of the driven rotor **20** relative to the drive rotor **10** as a rotor-to-rotor phase based on a rotational state of the control shaft **6**. The valve timing becomes timing appropriate for an operation condition of the internal combustion engine by the adjustment of the rotor-to-rotor phase.

Specifically, when the control shaft **6** rotates with the drive rotor **10** at the same speed, and thus the planetary carrier **40** does not rotate relative to the drive inner gear portion **12**, the geared rotor **30** does not undergo the planetary motion, and the geared rotor **30** rotates with the rotors **10** and **20**. As a result, the rotor-to-rotor phase and the valve timing is adjusted to be maintained. In contrast, when the control shaft **6** rotates at a low speed or in a reverse direction relative to the drive rotor **10**, and thus the planetary carrier **40** rotates in a cam retard direction relative to the drive inner gear portion **12**, the geared rotor **30** undergoes the planetary motion, and the driven rotor **20** rotates in the cam retard direction relative to the drive rotor **10**. As a result, the rotor-to-rotor phase and the valve timing is adjusted to be retarded. In contrast, when the control shaft **6** rotates at a speed higher than that of the drive rotor **10**, and thus the planetary carrier **40** rotates in a cam advance direction relative to the drive inner gear portion **12**, the geared rotor **30** undergoes the planetary motion, and the driven rotor **20** rotates in the cam advance direction relative to the drive rotor **10**. As a result, the rotor-to-rotor phase and the valve timing is adjusted to be advanced.

(Details of Configuration)

Hereinafter, the configuration of the valve timing apparatus **1** will be described in more detail.

As illustrated in FIG. **4**, in the ball bearing roller **60**, a plurality of ball bearings **66** are installed in a single row between an outer ring **62** and an inner ring **64**. An outer circumferential surface **62**_a_ of the outer ring **62** is coaxially fitted and mounted onto an inner circumferential surface **14**_a_ of the cylindrical portion of the cover member **14**. An inner circumferential surface **64**_a_ of the inner ring **64** is coaxially fitted on to support the supporting surface **46**. An inner circumferential surface **62**_b_ of the outer ring **62** and an outer circumferential surface **64**_b_ of the inner ring **64** function as raceway surfaces **62**_b_ and **64**_b_, respectively, which are rotatably in contact with the entirety of the ball bearings **66**. In the ball bearing roller **60**, the ball-shaped ball bearings **66** are interposed between the raceway surfaces **62**_b_ and **64**_b_ which are open on both ends **60**_a_ and **60**_b_ in the axial direction. That is, in the embodiment, the ball bearing roller **60** is an open end type single-row ball bearing.

As illustrated in FIG. **1**, a supply hole **70** is formed in the bottom wall portion of the driven rotor **20** to pass through the connection portion **22**. One end of the supply hole **70** communicates with a transportation hole **3** of the cam shaft **2**, and thus lubricant oil is supplied to the end of the supply hole **70** from a pump **9** driven by the crank torque. The other end of the supply hole **70** communicates with the inside of the drive rotor **10**, and thus the lubricant oil is supplied into the rotor **10** from the pump **9**. As a result, the lubricant oil supplied into the drive rotor **10** is introduced into the ball bearing roller **60** through an internal clearance and the like of the open end type planetary bearing **36** and the open end **60**_a_ located at a position at which the planetary bearing **36** is disposed.

In an initial state where the geared rotor **30** is not tilted as illustrated in FIG. **4**, one end surface **32**_a_ of the drive outer gear portion **32** faces open end surface **20**_a_ of the circumferential wall portion of the driven rotor **20** in the axial direction. In this case, a clearance **80** is formed entirely in the rotational direction (in the circumferential direction) between the end surfaces **32**_a_ and **20**_a_. In addition, in the initial state, the other end surface **32**_b_ of the drive outer gear portion **32** faces an end surface **14**_b_ of a flange portion of the cover member **14** in the axial direction. In this case, a clearance **82** greater than the clearance **80** is formed entirely in the rotational direction (in the circumferential direction) between the end surfaces **32**_b_ and **14**_b_. Furthermore, in the initial state, the radial loads Fc and Fu caused by the cam torque or the biasing force are exerted on the planetary carrier **40** in the first region **401**, and thus the planetary carrier **40** and the geared rotor **30** is tilted around a specific point R as a center point in the second region **402** as illustrated in FIGS. **5** and **6**. Since the geared rotor **30** is tilted relative to the axial direction, an outermost circumferential angled portion **32**_c_ (refer to FIG. **4** as well) on the eccentric side of the end surface **32**_a_ is elastically deformed and is brought into contact with the open end surface **20**_a_. That is, the outermost circumferential angled portion **32**_c_ of the end surface **32**_a_ of the geared rotor **30** is brought into elastic contact with the open end surface **20**_a_.

Here, as illustrated in FIG. **6**, a radius Ra is a distance between the outermost circumferential angled portion **32**_c_ that is brought into contact with the open end surface **20**_a_ by the tilting and a rotational center line O of the rotors **10** and **20**. A radius Rs is a distance between the supporting surface **46** and the rotational center line O. The radius Ra is set to be greater than the radius Rs. In addition, in the embodiment, when the outermost circumferential angled portion **32**_c_ is brought into contact with the open end surface **20**_a_, as illustrated in FIGS. **6** and **7**, the geared rotor **30** is set to be tilted together with the planetary carrier **40** and the inner ring **64** at an angle θ which is smaller than a maximum allowable angle of the inner ring **64** of the ball bearing roller **60**. When the outer ring **62** of the ball bearing roller **60** is fixed, the maximum allowable angle is an angle at which the inner ring **64** is tilted relative to the axial direction, and is at an angle at which the inner ring **64** is allowed to tilt in advance. For example, when the inner ring **64** moves relative to the outer ring **62** by only a predetermined amount of thrust, the maximum allowable angle is determined based on the fact that the ball bearings **66** between the inner ring **64** and the outer ring **62** are interposed between both ends in the axial direction of the raceway surfaces **62**_b_ and **64**_b_.

(Effects)

Hereinafter, effects of First Embodiment described above will be described.

In First Embodiment, the planetary carrier **40** receives the radial load in the first region **401** that is offset to one end **40***a* from the center position C in an axial direction, and the ball bearing roller **60** supports the planetary carrier **40** in the second region **402** that is offset to the other end **40***b* from the center position C. Accordingly, when the planetary carrier **40** receives the radial load in the first region **401**, the planetary carrier **40** is tilted around the specific point R in the second region **402**.

Here, the ball bearing roller **60** has a structure in which the plurality of ball bearings **66** are installed in a single row between the outer ring **62** that is mounted on the drive rotor **10** and the inner ring **64** that supports the planetary carrier **40**. Accordingly, in the single-row ball bearing roller **60**, when the planetary carrier **40** and the inner ring **64** is tilted relative to the axial direction, the tilt of the inner ring **64** is received (regulated) by the outer ring **62** through the ball bearings **66** in a single circumference, and thus effects of tilt regulation become small. At this time, the geared rotor **30** supported from the inside in the radial direction by the planetary carrier **40** also is tilted in the axial direction, and thus the geared rotor **30** is brought into contact with the driven rotor **20** in the axial direction. Since the effects of tilt regulation can be realized by the contact, the geared rotor **30** is unlikely to undergo gear rattle at the locations where the geared rotor **30** meshes with the driven inner gear portion **24**. However, at this time, the geared rotor **30** is tilted together with the planetary carrier **40** and the inner ring **64** at the angle θ which is smaller than the maximum allowable angle of the inner ring **64**. When the geared rotor **30** is tilted at the small angle θ, even in the single-row ball bearing roller in which both a total contact area between the outer ring **62** and the entirety of the ball bearings **66** and a total contact area between the inner ring **64** and the entirety of the ball bearings **66** are small, a large contact pressure can be prevented from occurring in a small area of a contact interface.

In this manner, wear can be prevented from occurring at the locations where the sloping geared rotor **30** meshes with the inner gear portions **12** and **24**. Wear can be also prevented from occurring in the ball bearing roller **60** the inner ring **64** of which is tilted, thereby durability of the entire apparatus **1** can be improved.

The radius Ra between the rotational center line O and the outermost circumferential angled portion **32***c* is set to be greater than the radius between the rotational center line O and the supporting surface **46** of the planetary carrier **40**. The outermost circumferential angled portion **32***c* in the eccentric side of the geared rotor **30** is brought into elastic contact with the driven rotor **20** due to the eccentricity of the geared rotor **30** with respect to the rotors **10** and **20**. Accordingly, a large contact area can be ensured, and thus the contact pressure can be reduced. As a result, wear of the contact interface can be suppressed. In contrast, in the single-row ball bearing roller **60** in which the inner ring **64** is supported by the supporting surface **46** having the radius Rs from the rotational center line O, which is smaller than the radius between the rotational center line O and the outermost circumferential angled portion **32***c*. Therefore, when the inner ring **64** is tilted together with the planetary carrier **40**, the amount of thrust Δ, which causes the inner ring **64** to be offset with respect to the outer ring **62** as illustrated FIG. **7**, can be reduced. Here, in particular, in the contact interface between the outer ring **62** and the ball bearings **66** and the contact interface between the inner ring **64** and the ball bearings **66**, wear caused by the offset of the inner ring **64** can be suppressed by the reduction of the amount of thrust Δ. The aforementioned configuration can contribute to an improvement in durability.

Furthermore, the lubricant oil from the pump **9** is supplied into the drive rotor **10**, and is introduced into the ball bearing roller **60** through the open end **60***a* in the axial direction between the outer ring **62** and the inner ring **64**. For this reason, during a cold temperature, the lubricant oil becomes resistant to rolling contact in the contact interface between the outer ring **62** and the entirety of the ball bearings **66** and in the contact interface between the inner ring **64** and the entirety of the ball bearings **66**. However, since the total contact area between the outer ring **62** and the entirety of the ball bearings **66** and the total contact area between the inner ring **64** and the entirety of the ball bearings **66** are small in the single-row ball bearing roller **60**, total resistance in the entire contact interface caused by the lubricant oil during a cold temperature can be reduced. Accordingly, the resistance of the lubricant oil is prevented from disturbing the rolling contact in the ball bearing roller **60**, thereby improving responsiveness of adjustment of the rotor-to-rotor phase or responsiveness of adjustment of the valve timing based on the planetary motion of the geared rotor **30**.

Furthermore, in the driven rotor **20** to which the cam torque is transmitted from the cam shaft **2**, the cam torque is converted at the locations where the driven inner gear portion **24** and the geared rotor **30** mesh with each other, and thus the radial load Fc is exerted on the first region **401** of the planetary carrier **40**. Accordingly, the planetary carrier **40** is tilted around the specific point R in the second region **402** by the exertion of the radial load Fc. However, when the geared rotor **30** is tilted together with the planetary carrier **40** and the inner ring **64** at the angle θ which is smaller than the maximum allowable angle of the inner ring **64** to be brought into contact with the driven rotor **20**, effects of the tilt regulation and effects of suppression of the contact pressure can be realized. Accordingly, even in the cam torque transmission structure that causes the radial load Fc, wear of the ball bearing roller **60** can be suppressed, thereby durability can be improved.

The pair of elastic members **50** installed between the geared rotor **30** and the planetary carrier **40** biases the geared rotor **30** toward the locations where the geared rotor **30** meshes with the inner gear portions **12** and **24**, and thus wear in each meshing location caused by gear rattle can be suppressed. The pair of elastic members **50** biases the planetary carrier **40** toward the side opposite to the geared rotor **30** by the biasing force, and the biasing force as the radial load Fu is exerted on the first region **401** of the planetary carrier **40**. Accordingly, the planetary carrier **40** is tilted around the specific point R in the second region **402**. However, when the geared rotor **30** is tilted together with the planetary carrier **40** and the inner ring **64** at the angle θ which is smaller than the maximum allowable angle of the inner ring **64** to be brought into contact with the driven rotor **20**, effects of the tilt regulation and effects of suppression of the contact pressure can be realized. Accordingly, even in a structure of biasing force being exerted, which causes the radial load Fu, wear of the ball bearing roller **60** can be suppressed, thereby durability can be improved.

In First Embodiment described above, the driven rotor **20** is equivalent to "a first rotor rotatably coupled with the cam shaft", and the driven inner gear portion **24** is equivalent to "a first gear portion". The drive rotor **10** is equivalent to "a second rotor rotatably coupled with the crank shaft", the drive inner gear portion **12** is equivalent to "a second gear portion", and the lubricant oil supplied into the drive rotor **10** from the pump **9** is equivalent to "lubricant liquid".

(Second Embodiment)

As illustrated in FIG. **8**, Second Embodiment of the present disclosure is a modification example of First Embodiment. In an initial state of the geared rotor **30** according to Second Embodiment, the end surface **32***a* faces an open end surface **2013***a* of a circumferential wall portion of a sprocket member **2013** of a drive rotor **2010** in the axial direction. In this case, an axial clearance **2080** is formed entirely in the rotational direction (in the circumferential direction) between the end surfaces **32***a* and **2013***a*. Here, similarly to in First Embodiment, the clearance **82** is formed between the end surfaces **32***b* and **14***b*, and is greater than the clearance **2080**. Accordingly, the radial loads Fc and Fu are exerted on the planetary carrier **40**, and thus the outermost circumferential angled portion **32***c* in the eccentric side of the end surface **32***a* of the geared rotor **30**, which is tilted together with the planetary carrier **40**, is elastically deformed and is brought into contact with the open end surface **2013***a*. That is, the outermost circumferential angled portion **32***c* of the end surface **32***a* of the geared rotor **30** is brought into elastic contact with the open end surface **2013***a*.

Even in Second Embodiment, the radius Ra between the outermost circumferential angled portion **32***c* and the rotational center line O is set to be greater than the radius Rs between the supporting surface **46** and the rotational center line O. When the outermost circumferential angled portion **32***c* is brought into contact with the open end surface **2013***a*, the tilt angle θ relative to the axial direction of the geared rotor **30** is set to be smaller than the maximum allowable angle of the inner ring **64**. Accordingly, a contact target of the geared rotor **30** is changed from the driven rotor **20** to the drive rotor **2010**, but the same effects described in First Embodiment can be realized.

(Other Embodiments)

The plurality of embodiments are described above, but the present disclosure is not limited to the embodiments. Various embodiments and combinations thereof can be applied insofar as the embodiments and the combinations do not depart from the scope of the present disclosure.

Specifically, Modification Example 1 may have a changed structure in which the driven rotor **20** rotates coupled with the crank shaft, and the rotor **10** rotates coupled with the cam shaft. In this case, the cam torque is transmitted to the rotor **10** as "the second rotor" from the cam shaft **2**.

In Modification Example 2, a ball bearing roller with the columnar rolling element **66** may be adopted as the single-row ball bearing roller **60**. Modification Example 3 may adopt a configuration in which the lubricant oil is not introduced into the ball bearing roller **60**. Here, particularly in Modification Example 3, for example, a closed end type radial bearing with both closed ends may be adopted as the single-row ball bearing roller **60**. Alternatively, Modification Example 3 may adopt a structure in which the lubricant oil is not supplied into the rotor **10**, or a structure in which the lubricant oil is supplied into the rotor **10**, but does not reach the ball bearing roller **60**.

Modification Example 4 may adopt a structure in which one elastic member **50** is provided or a plurality of the elastic members **50** are provided as far as the radial load Fu can be generated by the biasing force. In contrast, Modification Example 5 may adopt a structure in which the elastic member **50** is not provided.

In Modification Example 6, the radius Ra between the outermost circumferential angled portion **32***c* and the rotational center line O may be set to be smaller than or equal to the radius Rs between the supporting surface **46** and the rotational center line O. In addition to the embodiments and the modification examples which illustrate the apparatus that adjusts the valve timing of the intake valve, in Modification Example 7, the present disclosure may be applied to an apparatus which adjusts valve timing of an exhaust valve as the "valve" or which adjusts valve timing of both intake valve and exhaust valve.

What is claimed is:

1. A valve timing adjustment apparatus for adjusting valve timing of a valve, the valve being opened and closed by a rotation of a cam shaft, the cam shaft being rotated by torque transmitted from a crank shaft in an internal combustion engine, the valve timing adjustment apparatus comprising:

- a first rotor having a first gear portion and rotatably coupled with one of the crank shaft or the cam shaft;
- a second rotor having a second gear portion and interposing the first rotor on both sides in an axial direction, the second rotor rotatably coupled with an other of the crank shaft or the cam shaft;
- a geared rotor meshing with the first gear portion and the second gear portion and moving in planetary motion to adjust a relative phase between the first rotor and the second rotor;
- a planetary carrier supporting the geared rotor in a radial direction from an inside of the geared rotor, and receiving a radial load in a first region, the first region being offset to one end from a center position of the planetary carrier in the axial direction; and
- a ball bearing roller that is supported by the second rotor and supports the planetary carrier in the radial direction from an outside of the planetary carrier in a second region, the second region being offset to an other end from the center position of the planetary carrier in the axial direction, wherein
- the ball bearing roller is a single-row ball bearing roller that has an outer ring supported by the second rotor, an inner ring that supports the planetary carrier, and a plurality of ball bearings that are rotatably installed in a single row to be in contact with and between the outer ring and the inner ring,
- the geared rotor is tilted relative to the axial direction by the radial load and brought into contact with the first rotor or the second rotor in the axial direction, and
- an angle of the geared rotor relative to the axial direction is set to be smaller than a maximum allowable angle at which the inner ring is allowed to be tilted relative to the axial direction.

2. The valve timing adjustment apparatus according to claim **1**, wherein

- the geared rotor is eccentric with respect to the first rotor and the second rotor, and an outermost circumferential angled portion on eccentric side of the geared rotor is brought into elastic contact with the first rotor or the second rotor,
- the planetary carrier has a supporting surface coaxial with the first rotor and the second rotor and supports the inner ring with the supporting surface, and
- a radius between a rotational center line of the first rotor and the second rotor and the outermost circumferential angled portion is set to be greater than a radius between the rotational center line and the supporting surface.

3. The valve timing adjustment apparatus according to claim **1**, wherein

- the ball bearing roller has an open end between the outer ring and the inner ring that opens in the axial direction, and
- lubricant liquid supplied into the second rotor is introduced into the ball bearing roller through the open end.

4. The valve timing adjustment apparatus according to claim **1**, wherein a cam torque transmitted from the cam shaft to the first rotor or the second rotor is converted into the radial load at locations where the geared rotor meshes with the first gear portion or the second gear portion.

5. The valve timing adjustment apparatus according to claim **1**, further comprising:

an elastic member that is installed between the geared rotor and the planetary carrier to bias the geared rotor toward the locations where the geared rotor meshes with the first gear portion and the second gear portion, wherein

the radial load is generated by a biasing force by which the elastic member biases the planetary carrier toward a side opposite to the geared rotor.

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